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Article (Accepted Version)

Shafae, M, Mashouf, H, Sarmadian, A and Mohseni, S G (2016) Evaporation heat transfer and pressure drop characteristics of R-600a in horizontal smooth and helically dimpled tubes. *Applied Thermal Engineering*, 107. pp. 28-36. ISSN 1359-4311

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Accepted Manuscript

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PII: S1359-4311(16)31062-6

DOI: <http://dx.doi.org/10.1016/j.applthermaleng.2016.06.148>

Reference: ATE 8559

To appear in: *Applied Thermal Engineering*

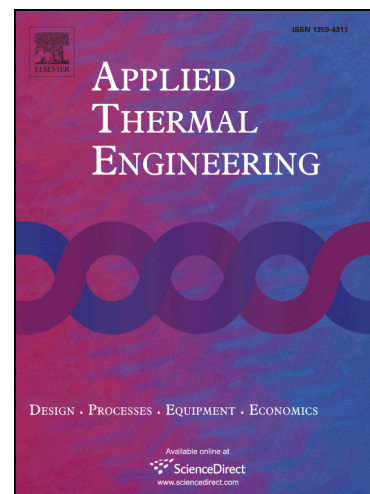
Received Date: 8 February 2016

Revised Date: 21 May 2016

Accepted Date: 21 June 2016

Please cite this article as: M. Shafaei, H. Mashouf, A. Sarmadian, S.G. Mohseni, Evaporation heat transfer and pressure drop characteristics of R-600a in horizontal smooth and helically dimpled tubes, *Applied Thermal Engineering* (2016), doi: <http://dx.doi.org/10.1016/j.applthermaleng.2016.06.148>

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Evaporation heat transfer and pressure drop characteristics of R-600a in horizontal smooth and helically dimpled tubes

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ABSTRACT

An experimental investigation was performed to evaluate the heat transfer and pressure drop characteristics of the hydrocarbon refrigerant R-600a during flow boiling inside a horizontal smooth tube with an inside diameter of 8.25 mm and a newly developed dimpled tube. The inner surface of the helically dimpled tube is enhanced by a newly modified pattern consists of both shallow and deep protrusions. The experimental tests were carried out varying: the refrigerant mass fluxes within the range of 155-470 kg/m²s; the vapor qualities up to 0.8; the constant heat flux of 15.8 kW/m² and saturation temperature of 56.5°C. Observations clearly indicate that the heat transfer performance is improved as tube's inner surface enhanced by this new pattern of protrusions. The experimental results show that the heat transfer coefficients of the dimpled tube are 1.29-2 times larger than a smooth tube with a pressure drop just ranging between 7% and

103% larger than the smooth tube. The highest enhancement in the heat transfer coefficient occurs at vapor quality of 0.2 and mass flow rate of 155 kg/m²s. On the other hand, the maximum increase of pressure drop takes place at vapor quality of 0.8 and mass flow rate of 305 kg/m²s.

Keywords: *Evaporation; Dimpled tube; R-600a; Heat transfer; Pressure drop.*

1. Introduction

Heat transfer enhancement has been an important consideration in the most heat transfer systems; especially for obtaining energy efficiency improvements in refrigeration and air conditioning applications. An evaporator is an important part of these systems, which determines the performance of the whole system. Therefore, the design of efficient evaporators is important to be worthy of attention. In connection with the point previously mentioned, there are two types of heat transfer enhancement techniques, active (external power required) and passive (no external power needed). Recently, several passive techniques for heat performance enhancement such as incorporating twisted tapes in the tubes, helical screw-tapes, rough surfaces, and dimples, have been reviewed by Liu and Sakr [1]. One of the interesting current passive techniques is the application of surface roughness. Enhanced inner surfaces are commonly used because they can produce higher heat transfer coefficient with a small pressure drop penalty. Moreover, tubes with inside roughness can reduce size and cost of the equipment and are successfully used in practical applications. In this regard, Li et al. [2] discussed how internal surface enhancement can efficiently improve heat transfer under conditions of mixed convection. However, they didn't investigate the influence of internal surface enhancement on two-phase flow. Kukulka and Smith

[3] evaluated the relationship between heat transfer enhancement and the surface geometry of tubes which was introduced by them. Surface enhancement of this tube is produced by VipertexTM and it has been named 1EHT. They compared the heat transfer for single phase flows and found that the 1EHT surface can produce heat transfer increases of more than 500% when compared to smooth tubes. Guo et al. [4] performed an experimental investigation to evaluate convective condensation and evaporation of R22, R32, and R410A inside a smooth tube, a herringbone tube and a newly developed enhanced surface EHT tube at low mass fluxes. The inner surface of the EHT tube was enhanced by dimple/protrusion and secondary petal arrays that it was produced by VipertexTM too. They found out that for condensation, the heat transfer coefficient of the EHT tube is 1.3-1.95 times larger than a smooth tube and the herringbone tube is 2.0-3.0 times that of the smooth tube. But they didn't investigate pressure drop of these tubes. Recently, Kukulka et al. presented experimental investigations of the outside [5] and inside [6] condensation and evaporation heat transfer of R410A, R22 and R32 that took place in a smooth tube and a VipertexTM 1EHT tube. For outside evaporation characteristics, average heat transfer coefficients for R22 and R410A on the 1EHT tube were in the range of one to four times greater than those of a smooth tube. They also demonstrated pressure drop results in both single and two-phase flows. Investigation of the influence of dimples/protrusion as a surface roughness on boiling are rarely reported in literature, but its influence on single phase flow are studied, extensively. Gupta and Uniyal [7] reported that the use of dimples over the heat exchanger tube surface can be very advantageous in relation to other enhancement methods due to the simplicity of their manufacturing process and no extra cost in raw materials or labor. Suresh et al. [8] performed an experimental investigation on the convective heat transfer and friction factor characteristics in the plain and helically dimpled tube in single-phase flow with constant heat

flux and using CuO/water nanofluid as working fluid. Their results revealed that the Nusselt number with dimpled tube and nanofluids under turbulent flow is about 19%, 27%, and 39% higher than the Nusselt number obtained with plain tube and dimpled tube. friction factors were about 2–10% higher than the plain tube. Recently, the application of shallow square dimples on the walls of flat tubes in compact heat exchangers for vehicular applications was experimentally investigated by Nascimento [9]. It was found that the use of shallow square dimples in flat tubes promoted an increase in the heat transfer augmentation factor between 1.37 and 2.28. These literature indicate the use of passive methods to enhance two-phase flow heat transfer performance based on the application of dimple/protrusion patterns, needs more investigation. In this regard, Li Ming et al. appraised the relationship between heat transfer enhancement and the surface geometry of a dimpled enhanced tube using experimental and numerical simulation techniques [10]. numerical simulations were conducted to simulate geometric design optimization of enhanced tubes and were validated with experimental data [11]. Results showed that dimples on tube surface present high heat transfer performance and compared to staggered configuration, the in-line dimples arrangement provided better overall heat transfer characteristics. In addition, the geometric parameters like the shape of dimple, depth, pitch, and starts were found to have significant effects on overall heat transfer performance while the dimple diameter has an insignificant effect on thermal performance. Therefore, the modified helically dimple pattern advantages based on the application of both deep and shallow dimples is the main concept of dimpled tube design which is used in the present study. Dimpled tube concept means reshaping plain tube, so evenly placed dimples (deep and shallow ones) in the tube wall form. Thus, at the first it can lead to swirling is created in the flow in areas close to the wall and finally produce separation of flow. On the other hand, it can produce a surface area

No previously reported data exists for the flow boiling inside the enhanced geometry which is considered in this study. On the other hand, the applications of hydrocarbons to replace conventional refrigerants in refrigeration and air-conditioning systems have been studied due to their zero ozone depletion potential and negligible global warming potential [13]. For instance, Comparing the ODP of R-600a (0) with R12 (1) [Molina and Rowland [14]] and 100 years GWP of R-600a (20) with R-134a (1370) [Kurylo [15]] shows that R-600a has a better environmental performance. In addition, Lee et al. in an experimental study [16] discussed that R-600a has a better energy performance compared with other refrigerants. Moreover, Chao-Chieh Yu and Tun-Ping Teng [17] indicated that the use of R600a can enhance the enhancement factors (EFs) of refrigerators. Therefore, the purpose of this study is to perform an experimental investigation of the convective evaporation heat transfer characteristics of hydrocarbon refrigerant R-600a, due to the environmental problems by CFCs and HCFCs and obvious necessity of development of new alternative refrigerants.

The schematic diagram of the experimental facility is shown in **Fig. 1**. The refrigerant loop consists of a test section, variable frequency gear pump, condenser, reservoir, pre-evaporators, thermocouples, Coriolis-effect mass flow meter and necessary instruments for flow control. The flow meter was a Coriolis mass flow meter with the accuracy within 0.1% of the full scale. The

gear pump with the nominal power of 20 L/min was coupled with the inverter to control the mass velocity. A differential pressure drop transducer apparatus with the accuracy of 0.075% of the full scale was installed to quantify the pressure drop through the test section. This apparatus is calibrated to measure the pressure drop in the range of 0-150 kPa. The flow pattern is visualized by a sight glass which has been set just after the test section. Electrical resistance heaters covered pre-evaporators. Pre-evaporators and test section were carefully insulated with 20 mm-thick glass wool to minimize heat loss through its walls. Pre-evaporators were installed just before the test section to achieve different vapor qualities at the inlet of the test section. Two RTD PT 100 type temperature sensors with accuracy of $\pm 0.1^\circ\text{C}$ with digital indicators and two pressure transducer was located just before and after the pre-heaters to determine the thermodynamic state and the enthalpy of the refrigerant at the inlet and outlet of the pre-evaporator then with the heat input, the quality of the refrigerant at the inlet of test section can be obtained by the energy balance equation. K-type thermocouples with a calibrated accuracy of $\pm 0.1^\circ\text{C}$ are located in 5 axial positions along the test evaporator. As can be seen from **Fig. 2** at each of this axial location, four thermocouples were located at four sides of the test tube to take account circumferential temperature variation. In order to measure the quasi-local heat transfer coefficient, wall temperature was considered as the average of all temperatures which was measured by the thermocouples. Thermocouples in the test evaporator were connected to a 24 channel data logger (Lutron-4208 SD) which can be used for K and T type thermocouples. Digital pressure indicator with the accuracy of 1 kPa was installed just before the test section to measure the inlet pressure of the test section. The saturation temperature at the average pressure of the test section was considered as the saturation temperature of the whole test section. In order to reach the steady condition and stabilize the operating condition and to ensure that the refrigerant was liquid

before entering the pump, a reservoir was installed between condenser and gear pump. The refrigerant loop was charged via the needle valve. Isobutane was used as the working fluid.

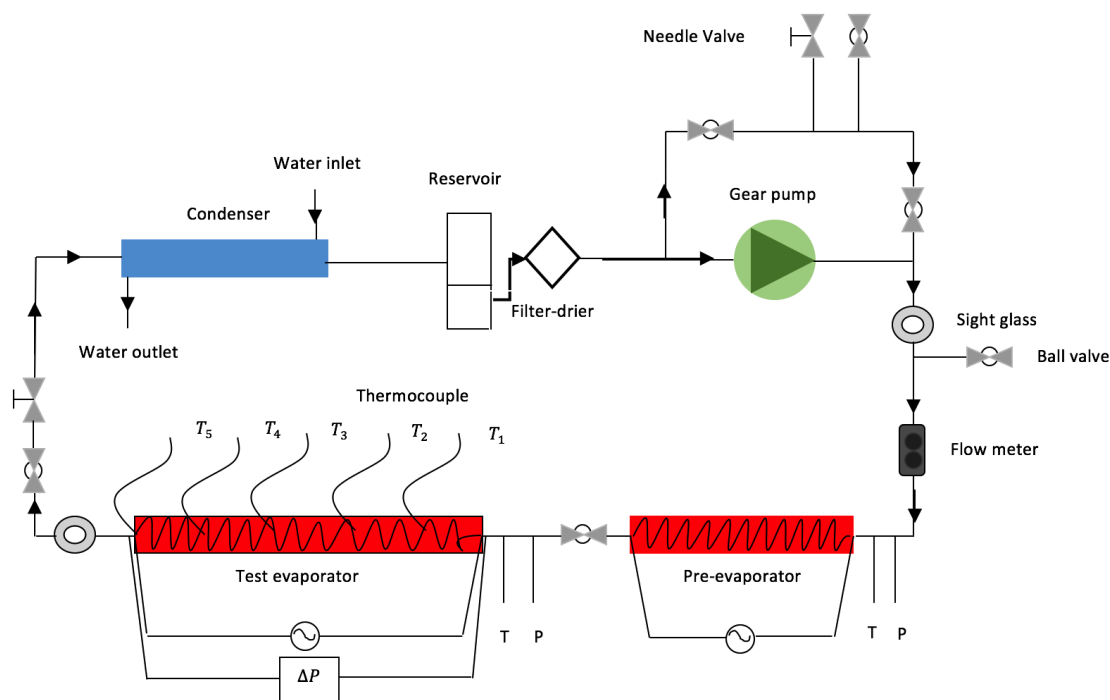


Fig. 1. Schematic diagram of experimental facilities

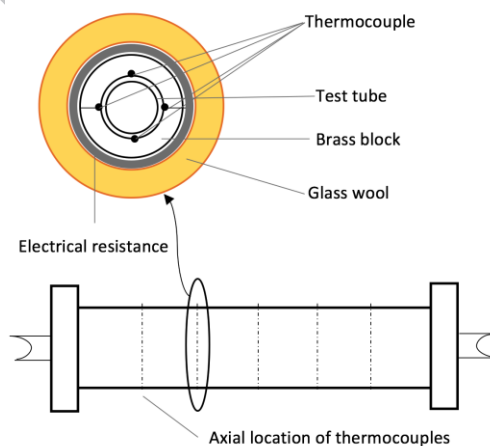


Fig. 2. Details of the test section

Two types of tubes were utilized as the test evaporator, one smooth tube and one newly designed tube that is enhanced by a kind of surface roughness. Parameters of the latter are shown in **Fig. 3**. The dimples were arranged helically with a pitch ratio ($= p/d$) of 1.21 on the test section. The diameters and depths of shallow dimples were maintained at a constant value of 1 mm and 0.5 mm, respectively. The diameters and depths of deep dimples were maintained at a constant value of 2 mm and 1 mm, respectively. Both of the tubes have an outer diameter of 9.5 mm, thickness of 0.6 mm, and length of 1000 mm. The range of operating parameters in the present study is given in **Table 1**.

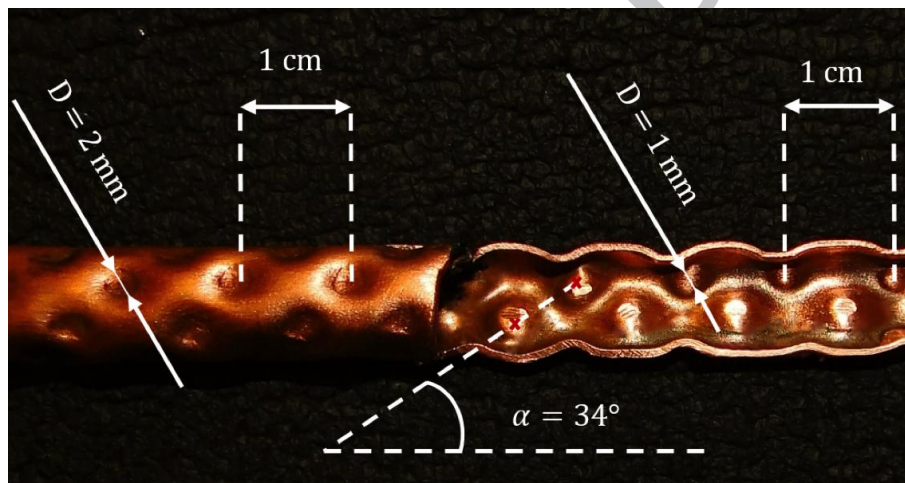


Fig. 3. Dimpled tube illustration and its pattern characteristics.

Table 1. Ranges of the operating parameters in the present study.

Parameter	Range
Refrigerant	R-600a
Inside diameter	8.25(mm)

The uncertainty analysis of the results was done by the method proposed by R. R. Schultz and R. Cole [18]. They suggested the following method of analyzing the effect of uncertainty in each variable on the uncertainty of the result:

where U_R is the estimate of the uncertainty in the calculated value of the desired variable R , due to the independent uncertainty U_{V_i} in the primary measurement of n number of variables, V_i , affecting the result. Applying Eq. (1), the uncertainty in the heat-transfer coefficient calculation can be written as:

It was found that the uncertainty in the determination of flow boiling heat transfer coefficient was within 8 percent for all the test runs. The uncertainties of experimental data and results are summarized in **Table 2**.

Table 2. Uncertainties of Primary measurements and experimental results.

Parameter	Uncertainties
Diameter	± 0.05 (mm)
Length	± 0.05 (mm)
Power	± 1 (W)
Temperature	± 0.1 °C
Vapor quality	± 0.005
Mass velocity	$\pm 1\%$ kg/m ² s
Pressure	± 1 (kPa)
Differential pressure	$\pm 0.05\%$
Heat transfer coefficient	$\pm 8\%$ W/m ² k
Frictional pressure drop	$\pm 4\%$

3. Data reduction

The experimental apparatus was installed to appraise the quasi-local heat transfer coefficient and frictional pressure drop during evaporation. It was assumed that the system reached to steady state condition when temperature and pressure were constant for at least 15 minutes. Then some test is iterated two times to check the reproducibility of test apparatus. The data is assembled for the flow of R-600a inside a smooth tube to establish the integrity of the experimental apparatus and to measure its performance against the performance of the dimpled tube, and then the data for dimpled tube was collected to quantify heat transfer coefficient and frictional pressure drop.

The efficiency of insulation of evaporators γ was defined as:

$$\gamma = \dot{m}_{ref}(h_3 - h_1)/(VI)_{pe} + (VI)_{ts} \quad (3)$$

Where \dot{m}_{ref} , V , I , h_3 , and h_1 are the refrigerant mass flow rate, electric voltage, electric current, enthalpy of the refrigerant at the test section outlet, and enthalpy of the refrigerant at the pre-evaporator inlet, respectively.

The quality of the refrigerant at the inlet and outlet of the test section were calculated from the following equations:

$$Q_{t,pe} = \gamma(VI)_{pe} = Q_{sens} + Q_{lat} \quad (4)$$

$$Q_{sens} = C_p \dot{m}_{ref}(T_{sat,pe} - T_{ref,pe,in}) \quad (5)$$

$$Q_{lat} = \dot{m}_{ref} h_{fg,pe} x_{in,ts} \quad (6)$$

By substituting Eq. (2) and Eq. (3) in Eq. (4) and rearranging the inlet quality of the test section can be expressed as:

$$x_{in,ts} = \frac{Q_{t,pe} - C_p \dot{m}_{ref}(T_{sat,pe} - T_{ref,pe,in})}{\dot{m}_{ref} h_{fg,pe}} \quad (7)$$

$$x_{out,ts} = x_{in,ts} + \gamma(VI)_{ts}/h_{fg,ts} \quad (8)$$

Where $Q_{t,pe}$, Q_{sens} , Q_{lat} , C_p , $T_{sat,pe,in}$, $T_{ref,pe,in}$, $h_{fg,pe}$, and $h_{fg,ts}$ are the total heat transferred in the pre-evaporator, sensible heat, latent heat, the specific heat of refrigerant was taken at average refrigerant temperature through the pre-evaporator, saturated temperature of refrigerant taken at

the average pressure of the pre-evaporator, temperature of the refrigerant at the inlet of the pre-evaporator, vaporization enthalpy of refrigerant at the average pressure of the pre-evaporator, and vaporization enthalpy of refrigerant at the average pressure of test section, respectively.

The local vapor quality through the whole test section was determined from:

$$x_{ts} = (x_{in,ts} + x_{out,ts})/2 \quad (9)$$

The quasi-local convective heat transfer coefficient h_{ref} was evaluated by:

$$\frac{1}{h_{ref}} = \frac{T_{wall} - T_{sat}}{q_x} - \frac{\ln(d_{out}/d_{inner})d_{inner}}{2k} \quad (10)$$

Where $q_x, T_w, T_b, d_{out}, d_{inner}$, and k are the heat flux, the wall temperature (the average temperature of all thermocouples), the bulk fluid temperature at the distance of x from the test section inlet, the outer diameter of the tube, the inner diameter of the tube, and thermal conductivity, respectively.

Pressure loss can be expressed by the following summation:

$$\Delta P_{tot} = \Delta P_{mom} + \Delta P_{fric} + \Delta P_{stat} \quad (11)$$

Where ΔP_{mom} , ΔP_{fric} and ΔP_{stat} are momentum pressure loss, frictional pressure loss, and static pressure loss, respectively. The Static part is equal to zero for a horizontal tube and, momentum pressure loss is calculated from the following correlation:

$$\Delta P_{mom} = \dot{m}_{ref}^2 \left\{ \left[\frac{(1-x)^2}{\rho_f(1-\alpha)} + \frac{x^2}{\rho_g\alpha} \right]_{out,ts} - \left[\frac{(1-x)^2}{\rho_f(1-\alpha)} + \frac{x^2}{\rho_g\alpha} \right]_{in,ts} \right\} \quad (12)$$

Where x and α are the vapor quality and void fraction, respectively. Void fraction can be calculated from the correlation proposed by Steiner[19] as follows:

$$\alpha = \frac{x}{\rho_g} \left\{ (1 + 0.12(1 - x)) \left(\frac{x}{\rho_g} + \frac{(1-x)}{\rho_f} \right) + \frac{1.18(1-x)[g\sigma(\rho_f - \rho_g)]^{0.25}}{G_{tot}^2 \rho_f^{0.5}} \right\}^{-1} \quad (13)$$

Finally, the frictional pressure loss can be achieved from the experimental data after calculating the momentum pressure loss.

4. Results and discussion

The results of heat transfer coefficients, pressure drop and performance comparison in both plain and dimpled tubes are discussed in separate sections as follows:

4.1. Heat transfer

The experimental heat transfer coefficients of refrigerant R-600a in the plain tube are measured and compared with those predicted by Gungor and Winterton [20]. As shown in **Fig. 4**, 72% of the data is contained inside 0 to -25% error windows. As mentioned by da Silva Lima in [21] the method of Gungor and Winterton [20] is very accurate in low heat transfer coefficients, typically constituted of slug flow pattern, but the under-prediction increases at high heat transfer coefficients, mainly constituted of annular flow. Therefore, all of the data which are shown in **Fig. 4** are related to the low vapor quality region which is related to stratified, intermittent, slug, and plug regions.

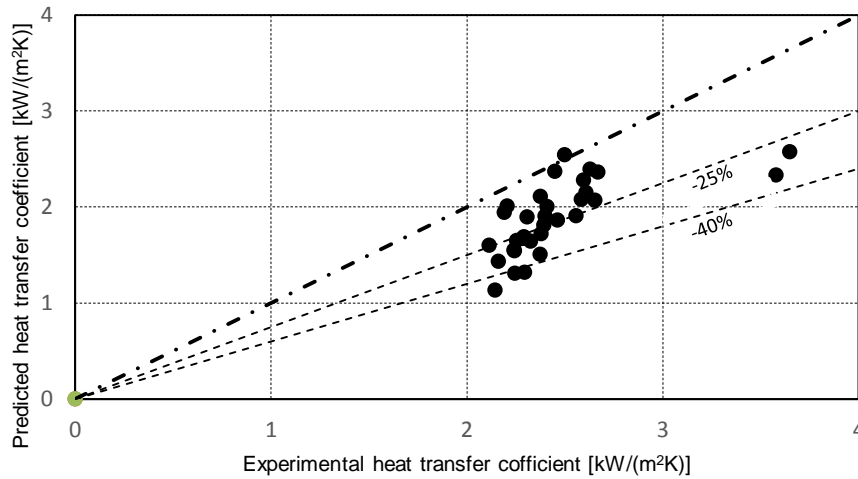
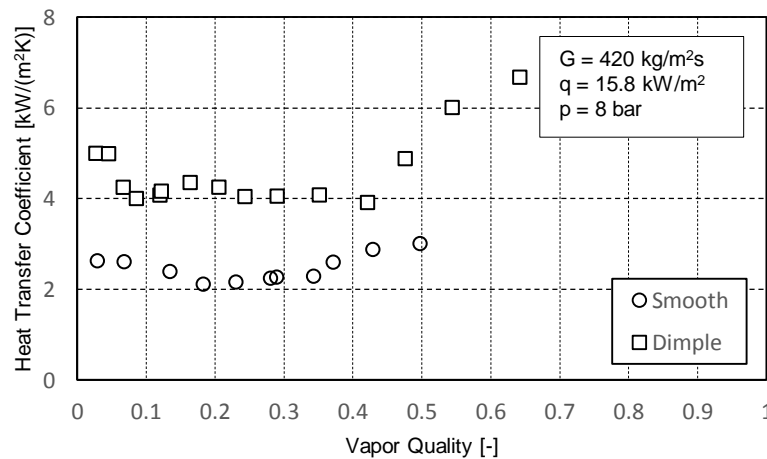


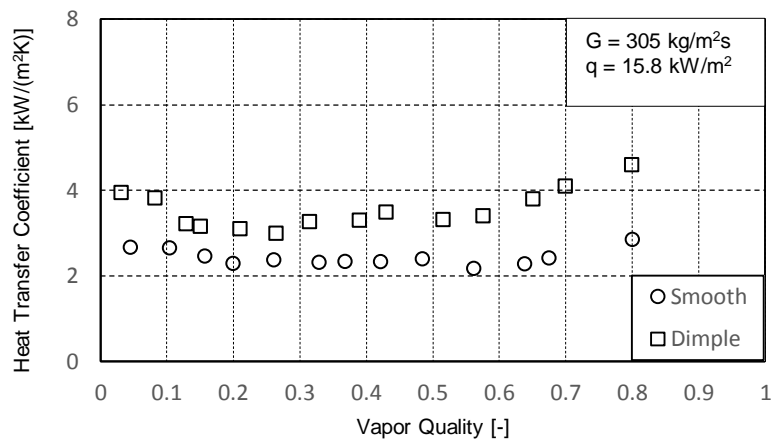
Fig. 4. Comparison of experimental and predicted evaporation heat transfer coefficient in the smooth tube.

Fig. 5 shows the variation of boiling heat transfer coefficient with vapor quality for four different mass fluxes. The pressure of the test section was kept constant at the average value of 8 bar for all vapor qualities. By inspection of **Fig. 5** it can be observed, there are two distinct heat transfer regions during evaporation in both of the smooth and dimpled tubes, decreasing heat transfer coefficient with vapor quality at low qualities and increasing at high qualities, presenting a local minimum in the vapor quality range between 15% and 45%. In the first one, nucleate boiling dominates. As vapor quality increases and annular flow appears, thinner liquid film (less thermal resistance) and high vapor velocity lead to an enhanced convection. Thus, the effective wall superheat is below the minimum value required for bubble nucleation on the wall and the active nucleation sites decrease. The effect of nucleate boiling declines and consequently, heat transfer coefficient decreases. These behaviors are very similar to the behaviors that are reported previously in [21]. For dimpled tubes, this region is smaller than smooth tubes and heat transfer coefficient increases at lower vapor qualities. Meanwhile, it presents higher heat transfer

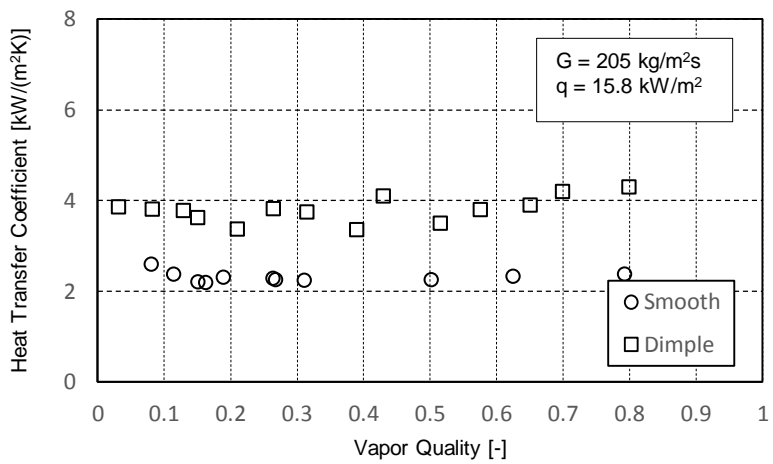
coefficient than smooth tubes. It's due to the protrusions' contribution to increasing superheated regions and more nucleation sites. In the second region, liquid convection dominates and as vapor quality grows the heat transfer coefficient increases. Actually, as refrigerant vaporizes during moving downstream, the void fraction increases and the refrigerant density declines. Hence, flow velocity grows that leads to enhancing liquid convection. For the dimpled tube, this region is wider than the smooth tube and heat transfer coefficient increases with a sharper gradient in comparison with the smooth tube. The heat transfer coefficient of the dimpled tube is 1.29-2.0 times that of the smooth tube. The highest and lowest heat transfer coefficient ratios are related to $x = 0.2$ and $\dot{m} = 155 \text{ kg/m}^2\text{s}$ and $x = 0.5$ and $\dot{m} = 420 \text{ kg/m}^2\text{s}$, respectively. This phenomenon is due to the increased turbulence, flow separation, boundary layer disruption, and flow mixing which are produced by protrusions.



(a)



(b)



(c)

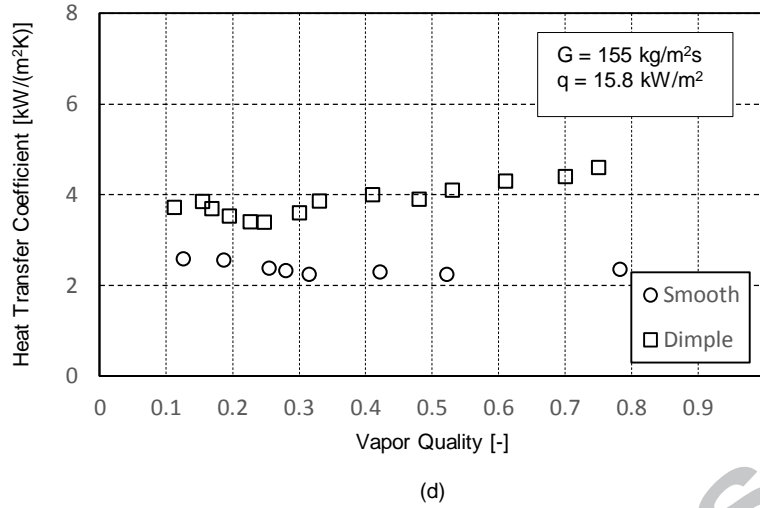


Fig. 5. Heat transfer coefficient under (a) $G = 420 \text{ kg/m}^2\text{s}$, (b) $G = 305 \text{ kg/m}^2\text{s}$, (c) $G = 205 \text{ kg/m}^2\text{s}$ and (d) $G = 155 \text{ kg/m}^2\text{s}$ all with $q = 15.8 \text{ kW/m}^2$ and $P = 8 \text{ bar}$

To illustrate the influence of mass flux on heat transfer for both of the smooth and dimpled tubes, the variation of heat transfer coefficient with vapor quality for two different mass fluxes of $155 \text{ kg/m}^2\text{s}$ and $420 \text{ kg/m}^2\text{s}$ is shown in **Fig. 6**. It's clear that by increasing the mass flux, the refrigerant velocity increases and therefore liquid convection and heat transfer coefficient enhances. **Fig. 6** depicts that in the high quality region corresponding to the annular regime, where convective boiling is dominant, the influence of mass flux became stronger as stated in [22] and heat transfer coefficients diverge from each other as mass flux increases. On the contrary, where vapor quality is low, nucleate boiling is dominant and mass flux has negligible influence and heat transfer coefficients tend to merge together. By considering in **Fig. 6** it can be understood that for both of the tubes, as mass flux and vapor quality increase, the heat transfer coefficients soar at a higher rate and this phenomenon is more significant for the dimpled tube.

The variation of heat transfer coefficient with Reynolds number for both of the dimpled and smooth tubes at two different constant vapor qualities $x = 0.173$ (related to the region that nucleate boiling is dominant) and $x = 0.45$ (related to the region that convective boiling is dominant) has been shown in **Fig. 7**. As can be seen, in all range of tested Reynolds number the heat transfer coefficient of the dimpled tube is around 1.3-2.0 times that of the smooth tube at $x = 0.173$ and is 1.45-2.0 times that of the smooth tube at $x = 0.45$. The heat transfer coefficient behaves non-monotonically with Reynolds number and discussions provided in [22] might be used to explain this behavior. The enhancement by interfacial turbulence is more significant at low Reynolds number and tends to be lower at high Reynolds number. This is due to enhanced turbulence that already exists in high Reynolds number. In addition, it has been previously explained in [23] and [24] that the transfer of momentum by the smaller eddies is more efficient than the transfer of momentum by larger eddies. Therefore, at high Reynolds number of flow, the heat transfer coefficient enhancement is lower than heat transfer coefficient enhancement at low Reynolds number since as Reynolds number decreases, the size of turbulent eddies decreases too.

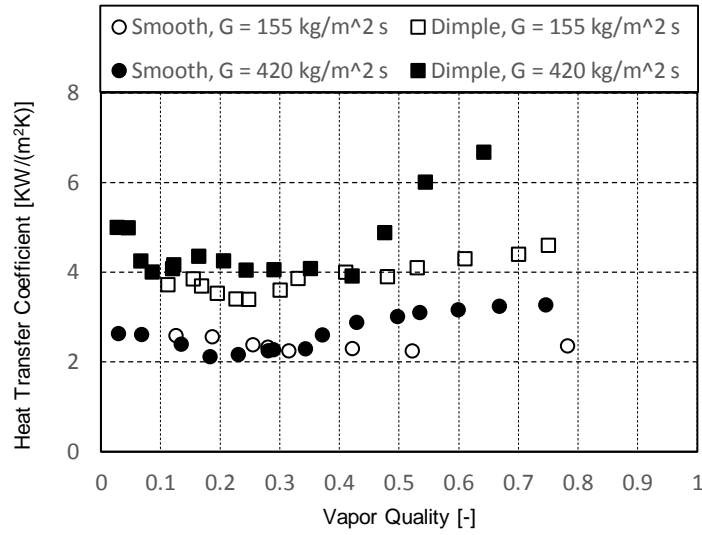


Fig. 6. The variation of heat transfer coefficient with vapor quality for two different mass fluxes of $155 \text{ kg/m}^2\text{s}$ and $420 \text{ kg/m}^2\text{s}$.

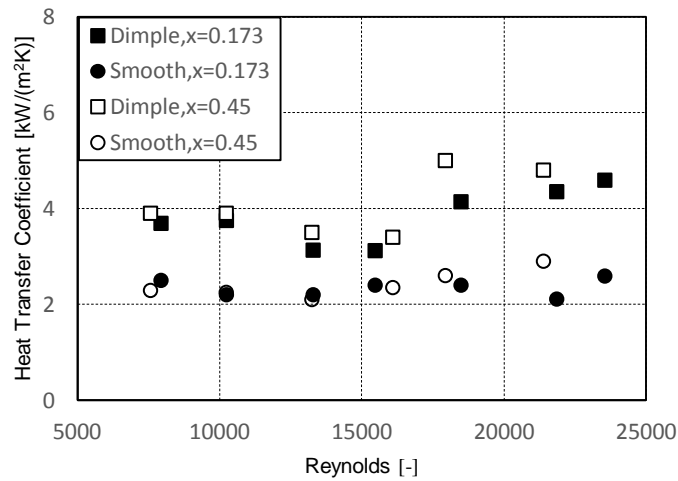


Fig. 7. The variation of heat transfer coefficient with Reynolds number for both Dimpled and smooth tubes at two different constant vapor qualities $x = 0.173$ (related to the region that nucleate boiling is dominant) and $x = 0.45$ (related to the region that convective boiling is dominant).

4.2. Pressure drop

First of all the pressure loss results of the plain tube with those predicted by Friedel [25] are compared. For this purpose, pressure loss was measured for seven different mass velocities (155-470 $\text{kg/m}^2\text{s}$) and at each of mass velocities, vapor quality varied within the range of 0.1-0.8. As depicted in **Fig. 8**, there is a good agreement between the obtained experimental data and the values predicted; 87% of the data is contained inside +10% to -25% error window and just 13% is contained inside -25% to -40% error window.

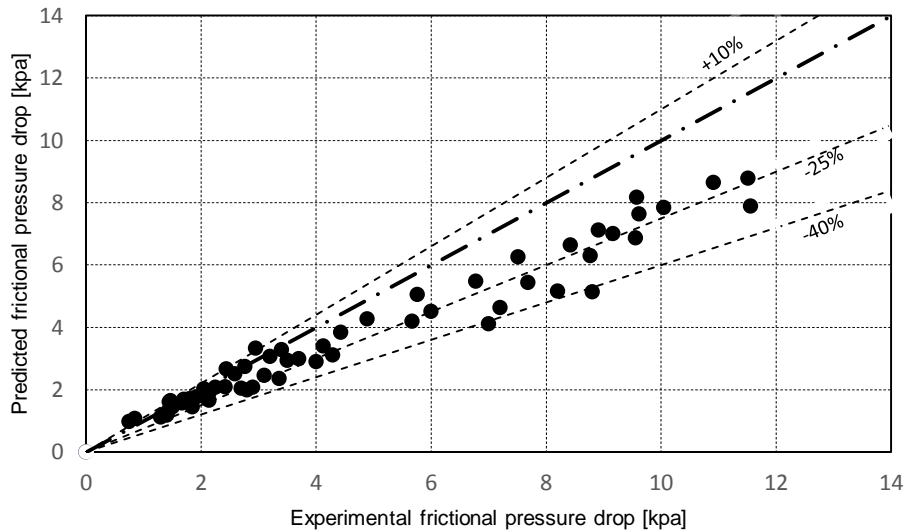
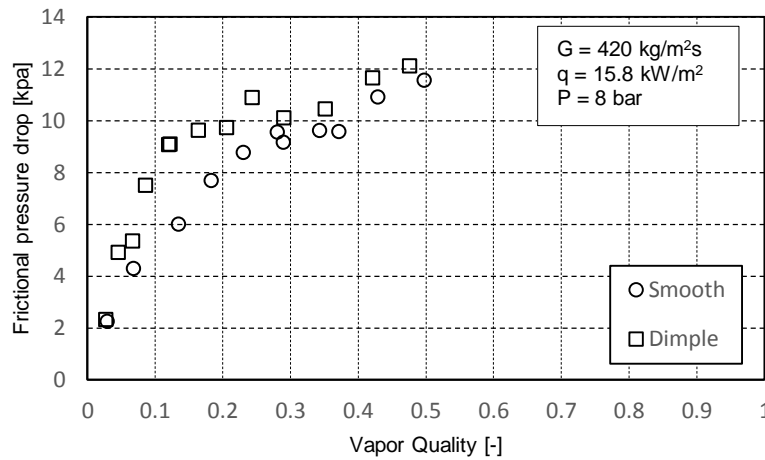


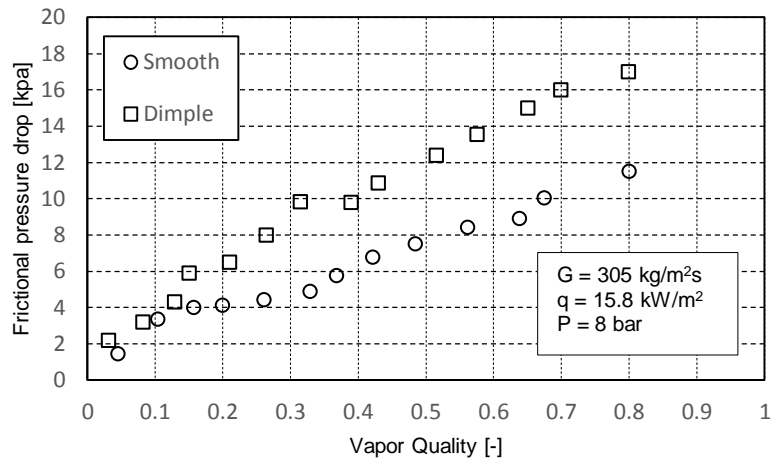
Fig. 8. Comparison of experimental and predicted frictional pressure drop in the smooth tube during evaporation.

The frictional pressure drop as a function of vapor quality for both of the dimpled and smooth tubes at four different mass fluxes 420 $\text{kg/m}^2\text{s}$, 305 $\text{kg/m}^2\text{s}$, 205 $\text{kg/m}^2\text{s}$ and 155 $\text{kg/m}^2\text{s}$ are illustrated in **Fig. 9**. The frictional pressure drop is evaluated through the test section with the length of 1 meter. For all of the mass fluxes, it can be observed that owing to the increasing interaction between the gas and liquid phases, the frictional pressure drop increases with the vapor quality and at a constant mass flux this is due to the fact that, as the quality of vapor inside

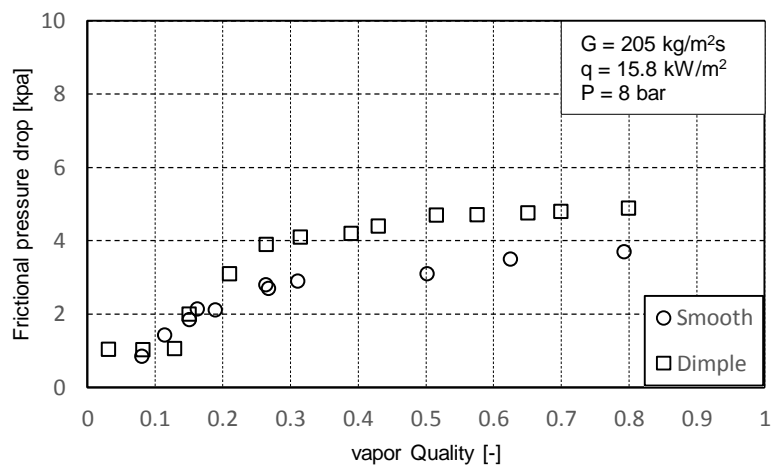
the tube grows; the velocity of fluid increases which in its own turn leads in more shear stress and more pressure loss. This phenomenon happened with the same gradient for both the dimpled and smooth tubes. For mass fluxes of $420 \text{ kg/m}^2\text{s}$, $305 \text{ kg/m}^2\text{s}$, $205 \text{ kg/m}^2\text{s}$ and $155 \text{ kg/m}^2\text{s}$, the pressure drop of the dimpled tube is 1.1-1.44, 1.07-1.7, 1.27-1.47 and 1-2.03 times that of the smooth tube, respectively. This can be contributed to the fact that, the use of dimples increases the frictional surface which results in more frictional pressure loss. For all of the mass velocities, the minimum pressure drop is related to minimum vapor quality and the maximum pressure drop is related to maximum vapor quality.



(a)



(b)



(c)

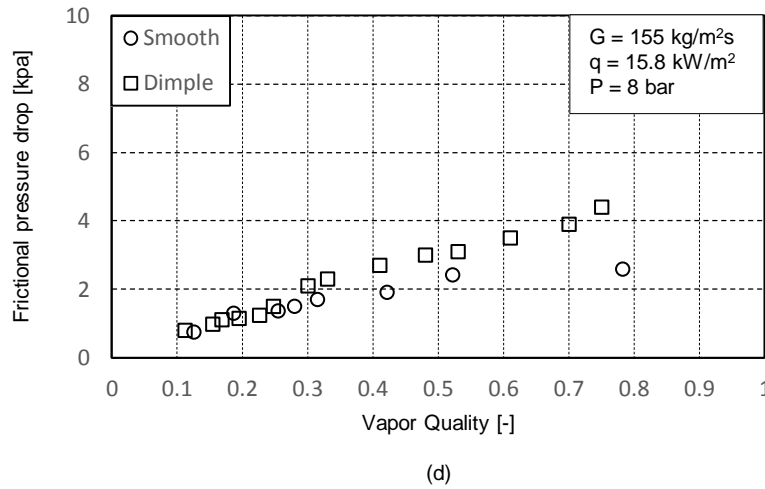


Fig. 9. Frictional pressure drop under (a) $G = 420 \text{ kg/m}^2\text{s}$, (b) $G = 305 \text{ kg/m}^2\text{s}$, (c) $G = 205 \text{ kg/m}^2\text{s}$, and (d) $G = 155 \text{ kg/m}^2\text{s}$ all with $q = 15.8 \text{ kW/m}^2$ and $P = 8 \text{ bar}$

To illustrate the influence of mass flux on frictional pressure drop for the dimpled tube, the variation of frictional pressure drop with vapor quality for two different mass fluxes of $155 \text{ kg/m}^2\text{s}$ and $420 \text{ kg/m}^2\text{s}$ is shown in **Fig. 10**. As it was stated previously [26], the frictional pressure drop is a change of pressure resulting from the energy dissipated in the flow by friction, eddying etc. As the flow proceeds downstream and vaporization takes place, the density of liquid-vapor mixture decreases. As a result, the flow accelerates and the frictional pressure drop increases. The frictional pressure drop of dimpled tube for $G=420 \text{ kg/m}^2\text{s}$ is 2.9-7.31 times that of the $G=155 \text{ kg/m}^2\text{s}$. In the vapor quality range below the $x=0.3$, increasing frictional pressure drop with vapor quality for $G=420 \text{ kg/m}^2\text{s}$ takes place with steeper gradient than

$G=155\text{ kg/m}^2\text{s}$. This is due to the turbulence promotion that is caused by dimples which lead to increased frictional pressure loss.

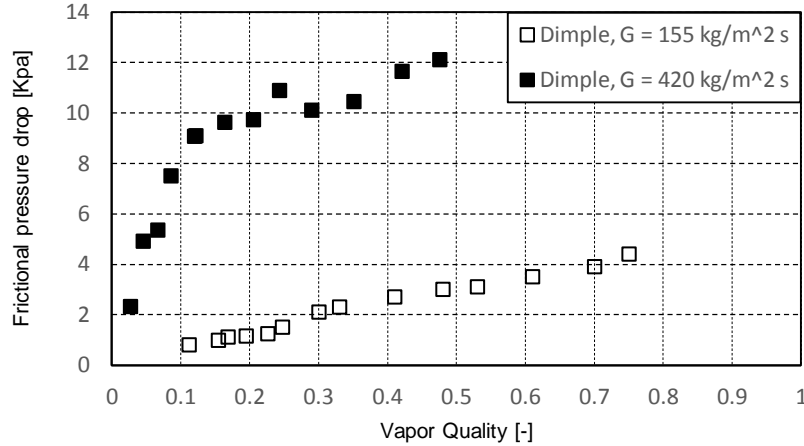


Fig. 10. The variation of frictional pressure drop with vapor quality for two different mass fluxes of $155\text{ kg/m}^2\text{s}$ and $420\text{ kg/m}^2\text{s}$.

The variation of frictional pressure drop with liquid film Reynolds number for both the dimpled and smooth tubes at two different constant vapor qualities $x = 0.173$ and $x = 0.45$ has been shown in **Fig. 11**. In all range of tested Reynolds numbers, the frictional pressure drop of the dimpled tube is 1.25-1.92 times that of the smooth tube at $x = 0.173$ and 1.11-1.87 times that of the smooth tube at $x = 0.45$. Frictional pressure drop's behavior with Reynolds number is nearly same for the dimpled and smooth tubes and for all of the tested vapor qualities. For dimpled tube at vapor quality of $x = 0.45$, frictional pressure drop values are about 1.3-2.7 times that of those at $x = 0.173$. In general, as the vapor quality increases, the velocity of fluid grows and leads to increase shear stress and frictional pressure drop.

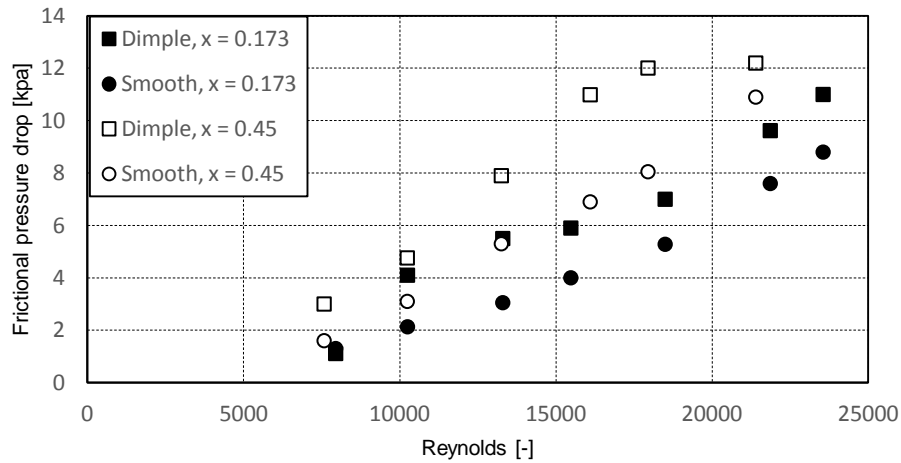


Fig. 11. The variation of frictional pressure drop with liquid film Reynolds number for both Dimpled and smooth tubes at two different constant vapor qualities $x = 0.173$ and $x = 0.45$.

4.3. Performance comparison

As mentioned before, the application of dimples in tube wall can produce a thermal surface area increase. In this regard, a performance factor PFS is defined by Guo et al.[4] as the ratio of the average heat transfer coefficient of the enhanced tube to that of nominal plain tube with consideration of the area enhancement ratio, as given by Equation bellow:

$$PFS = \frac{h_D}{h_p} \cdot \frac{A_p}{A_D} \quad (14)$$

As can be seen from **Fig. 12** for all tested Reynolds numbers, PFS values are larger than unity. It indicates that the heat transfer enhancement ratio is larger than the inner surface area ratio, so dimpled tube in addition to increasing surface area, it enhances heat transfer coefficient by flow separation, boundary layer disruption, secondary flow generation, increased interfacial turbulence and more nucleation sites.

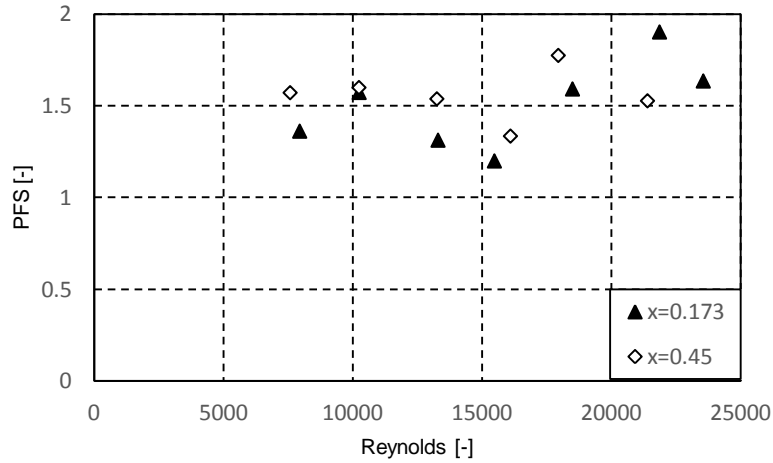


Fig. 12. *PFS* performance factor for the Dimpled tube.

In other consideration, analysis of the present experimental data showed that the use of dimples/protrusions as inner surface enhancement of the horizontal tubes increases the heat transfer coefficient with a penalty of pressure drop increasing. Heat transfer enhancement with a relatively high pressure loss is not interesting. Obviously, the heat transfer coefficient and pressure drop are independent parameters which don't relate by an equation. Therefore, the comparison needs a third parameter which relates to both of them and can produce the suitable condition for this purpose. In this regard, Agrawal and Verma [27] proposed the ratio of pumping power to enhanced heat transfer rate or enhanced heat transfer coefficients as an alternative to be used as the criterion of performance evaluation. The increased pumping power due to the presence of dimples can be calculated by multiplying the volumetric flow rate and produced pressure drop in the test evaporator. Therefore, the pump power can be calculated from following equation:

$$\dot{w} = \dot{v}\Delta P \quad (15)$$

$$\frac{(\frac{\dot{v}\Delta P}{h})_p}{(\frac{\dot{v}\Delta P}{h})_D} = \frac{(\frac{h_D}{h_p})}{(\frac{(\Delta P)_D}{(\Delta P)_n})} = \frac{R_h}{R_{\Delta P}} \quad (16)$$

The variation of *PFE* versus Reynolds number at two constant vapor qualities $x = 0.173$ and $x = 0.45$ is shown in **Fig. 13**. No consistency observed for the variation of *PFE* with Reynolds number. As can be seen, about the 60% of *PFE* values are larger than unity which is desirable. Flows with Reynolds number higher than 18000 have the best performance compared to other Reynolds numbers. Also, it is concluded that at Reynolds number lower than 18000, flow with vapor quality of $x = 0.45$ has a better performance than flow with vapor quality of $x = 0.173$. As a conclusion, employing the dimpled tube at high vapor qualities corresponding to annular regimes is recommended.

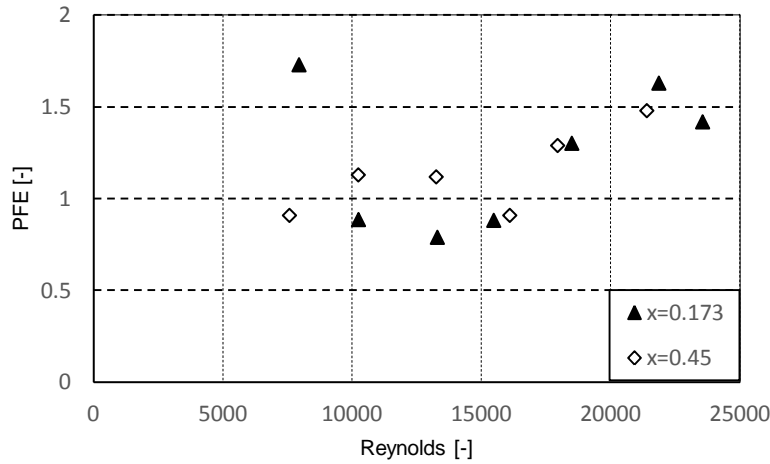


Fig. 13. *PFE* performance factor for the Dimpled tube.

5. Conclusions

An experimental apparatus has been set up for evaluating the heat transfer coefficient and pressure drop of natural refrigerant of R-600a as a function of vapor quality and Reynolds number during evaporation. Two test sections were used. One smooth copper tube and one newly designed copper tube that was enhanced by two kinds of protrusions in the inner side of the tube. Test sections were uniformly heated by the Joule effect. The experimental tests are carried out by varying the vapor quality within the range of 0 - 0.8 and the refrigerant mass flux within the range of 155 - 470 kg/m^2s .

The dimpled tube augmented the heat transfer coefficient up to 100% and up to 29% above the smooth tube in the best condition and worst condition, respectively. Experimental results showed that at all mass fluxes for the dimpled tube, the region which heat transfer coefficient grows with positive gradient is wider than that of the smooth tube. Meanwhile, it was found that

the heat transfer coefficient ratios (R_h) did not depend on mass flux and nearly were the same for all mass fluxes.

By investigation in pressure drop data, it can be discovered that the frictional pressure drop of the dimpled tube is always higher than those of plain tube by a factor ranged from 7% to 103%. The worst condition occurred at vapor quality of 0.78 and mass flux of $155 \text{ kg/m}^2\text{s}$. Moreover, pressure drop ratios ($R_{\Delta P}$) were not influenced by mass flux and nearly had same values at all mass fluxes.

The PFS values are always larger than unity and it indicates that in addition to increasing inner surface area ratio, flow separation, boundary layer disruption, secondary flow generation, increased interfacial turbulence, and more nucleation sites contribute to enhancing heat transfer coefficient. On the other hand, about 60% of PFE values are larger than unity. This outcome shows that the performance of Dimpled tube is desirable.

Nomenclature

\dot{m}	mass flow rate (kg/s)
A	surface area (m^2)
d	diameter (mm)
G	mass velocity ($\text{kg/m}^2\text{s}$)
h	specific enthalpy (kJ/kg)
h	heat transfer coefficient ($\text{kW/m}^2\text{K}$)
I	current (A)
K	thermal conductivity (W/m.K)
L	length (mm)

P	pressure (kPa)
PFE	energy performance factor
PFS	surface performance factor
Q	rate of heat transfer (W)
q	heat flux (kW/m ²)
Re	Liquid film Reynolds number
	$Re = \frac{4G(1-x)\delta}{(1-\varepsilon)\mu_L}$
T	temperature (K)
V	voltage (V)
x	vapor quality
p	pitch ratio

Greek symbols

\dot{v}	volumetric flow rate (m ³ /s)
ΔP	pressure drop (kPa)
α	void factor
γ	efficiency factor
ρ	density (kg/m ³)
σ	surface tension (N/m)
ϖ	power (W)
μ	viscosity (kg/m.s)
δ	liquid film thickness

ε void fraction of vapor

Subscripts

b Bulk

f liquid phase

$fric$ Frictional

g vapor phase

lat Latent

mom Momentum

p Plain

pe pre-evaporator

ref Refrigerant

s Smooth

sat saturated

sen Sensible

tot Total

ts test section

w Wall

L Liquid

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Highlights

- Heat transfer coefficient and pressure drop in dimpled tube are considered together.
- A local minimum was observed in h - x chart at low vapor qualities for all mass fluxes.
- The use of dimpled tubes will improve the heat transfer augmentation factor.
- Performance factors (PFE & PFS) showed that the use of dimpled tube is beneficial.